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# Effects of water vapor condensation on the convection heat transfer of wet flue gas in a vertical tube

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**Abstract**

The annular thin film condensation of water vapor in wet flue gas flowing through a vertical tube was studied theoretically and experimentally. Especially discussions were conducted on the effects induced by condensation of small amount of water vapor (10–20% fraction) on convection heat transfer in a vertical tube. The convection heat transfer was enhanced by the condensation of the condensable gas (water vapor) existing in the wet flue gas. The experiments also indicate that the wall temperature was an important factor affecting the condensing rate and the fog formation in the wet flue gas. The fog formation in the wet flue gas has a significant influence on the condensing rate and associated heat transfer. The proportion of the sensible and latent heat contribution to total heat transfer would change with the *Re* number. When the wall temperature is much lower than the saturation temperature of the inlet wet flue gas, the effect of the superheat could be neglected. But when the fog forms in the bulk flow, the temperature and concentration profile would be altered. 

*Keywords:* Condensation; Wet flue gas; Convection

## 1. Introduction

The annular thin film condensation of vapor inside a vertical tube is one of the important processes in the chemical and power industry and has been extensively studied [1–3]. The pioneering work of Nusselt was recently modified to include the effect of interfacial shear stress on both condensation flow and the characteristic of vapor velocity diminishing along the length of the tube [4]. All these investigations were concerned with condensation of pure saturated vapors.

In practical operations of tube condensers, some amount of non-condensable gas may exist in working vapors. It was well recognized that existence of non-condensable gas in vapors could greatly reduce condensation heat transfer and deteriorate the performance of devices. Many investigations were conducted on this topic [5].

But for condensing boiler and latent heat recovery from wet flue gas, the water vapor volume fraction is only about 10–20%. Predicting the effects of water vapor condensation in the wet flue gas on convection heat transfer in vertical tube seems to be very important for practical applications and displays special theoretical interest.

For condensation of vapor with non-condensable gas when vapor fraction is small the mixed gas flow is not to be decelerated obviously and non-condensable gas maintains the bulk flow. These features preclude the using of the theoretical results obtained through investigating the effects of non-condensable gas on condensation. Therefore, a new analysis which should consider these two factors simultaneously is highly necessary for condensation of a gas–vapor mixture with low fraction of water vapor flowing along tubes. Although the condensation of vapor from a vapor–gas mixture in tubes was studied historically [6,7], the main interest of these literatures was not to detect the effects of condensation of vapor on convection heat transfer, but to predict the

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**Nomenclature**

| | |
|---|---|
| $d$ | diameter (m) |
| $g$ | mass transfer coefficient (kg/m$^2$ s) |
| $h$ | heat transfer coefficient (W/m$^2$ K) |
| $k$ | thermal conductivity (W/m K) |
| $L$ | latent heat of condensation (J/kg), length (m) |
| $Le$ | Lewis number |
| $M$ | mass flux (kg/m$^2$ s) |
| $m$ | mass flux (kg/s) |
| $Nu$ | Nusselt number |
| $Pr$ | Prandtl number |
| $Q$ | heat flux (W/m$^2$ K) |
| $Re$ | Reynolds number |
| $Sc$ | Schmidt number |
| $Sh$ | Sherwood number |
| $T$ | temperature (K) |
| $t$ | temperature (°C) |
| $u$ | gas bulk velocity (m s) |
| $v$ | induced velocity (m/s) |
| $x, y$ | coordinates (m) |

*Greek symbols*

| | |
|---|---|
| $\Theta$ | correct factor |
| $\delta$ | thickness (m) |
| $\mu$ | viscosity (kg/m s) |
| $\phi$ | dimensionless mass flux |
| $\rho$ | density (kg/m$^3$) |

*Subscripts*

| | |
|---|---|
| c | condensation |
| g | gas |
| i | interface |
| l | liquid |
| m | mass |
| n | non-condensable |
| t | heat |
| v | vapor |
| w | wall |

effects of non-condensable gas on condensation heat transfer. For the former case (low vapor portion), the search of literature has revealed that there is a great shortage of investigations and no available works published indicate what is a remarkable increment in the associated convection heat transfer in tubes induced by vapor condensation of wet flue gas except a few reports on the transport phenomena of wet air [8]. Thus the topic is wide open for study to understand the convection associated with low fraction vapor condensation. The aim of this paper is to present some experimental results to illustrate and develop a theoretical model to examine the effects of condensation of a gas–vapor mixture on total heat transfer in a vertical tube.

## 2. Experimental

A diagrammatic sketch of the experimental facility used in conducting condensation experiments of wet flue gas is shown in Fig. 1. The vertical test condenser was composed of an inner stainless steel tube having outer diameter 8 mm and inner diameter 6 mm, concentrically surrounded by a cooling jacket. The length of the condensing section was 0.475 m.

Thermocouples were used to measure tube wall and wet flue gas temperatures at 0.15 m intervals along the length of the condenser. The wall temperatures were measured by means of thermocouples embedded in the tube wall as shown in Fig. 2. The wet flue gas tem-

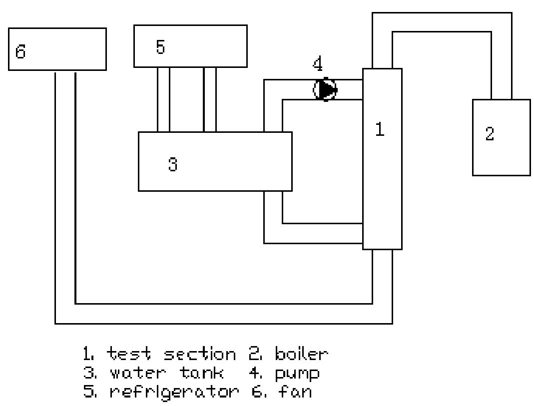


Fig. 1. Diagrammatic sketch of the test system.

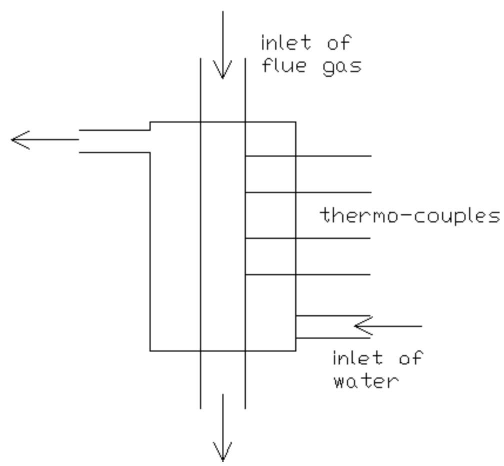


Fig. 2. The test section.

peratures at different locations were measured by thermocouples suspended in the vertical tube.

## 3. Model

### 3.1. Description

For condensation of a vapor–gas mixture a lot of research was reported in the open literature [9]. The film model proposed by Colburn and Hougen [6] is the simplest physical description of the mass transfer in a mixture.

Owing to complex physical characteristics of wet flue gas and condensing liquid, the gas–vapor mixture was treated as an air–vapor mixture. According to the assumption accepted for film model, the heat, mass and momentum transfer occur through the condensation film and gas layer. For a wet flue gas exhausted from natural gas boiler the vapor fraction is very low and the liquid film would be very thin. The main thermal resistance is in the gas layer.

Consider steady, laminar, filmwise condensation of a vapor–gas mixture in a vertical tube with down-flow. The mixture flow entering the tube is fully developed. The tube surface temperature is kept constant. Fig. 3 illustrates a physical model schematically.

According to the energy conservation at the interface between condensing liquid and vapor–gas mixture, energy equation is given by

$$\frac{k}{\delta}(t_\mathrm{i} - t_\mathrm{w}) = h_\mathrm{g}(t_\mathrm{g} - t_\mathrm{i}) + mL. \tag{1}$$

The first-term of the right-hand side represents the convection heat transfer between mixture bulk flow and condensing liquid surface, and the second-term is the condensation heat transfer.

When condensing rate is zero the transport process would be complete convection heat transfer. For pure vapor condensation the convection heat transfer can be neglected and Eq. (1) will be reduced to the Nusselt theory.

The relation between liquid film thickness and condensing liquid flow rate can be obtained from liquid mass conservation

$$\frac{\mathrm{d}M}{\mathrm{d}x} = m\pi(d - 2\delta). \tag{2}$$

For wet flue gas the water vapor concentration is generally 0.1–0.13 [10] and the fraction of condensed water vapor is only about 0.2–0.3, so the condensing film will be very thin and Eq. (2) can be simplified as

$$\frac{\mathrm{d}M}{\mathrm{d}x} = m\pi d. \tag{3}$$

From Nusselt's condensation theory, the liquid velocity could be derived as

$$u_\mathrm{l} = \frac{1}{\mu_\mathrm{l}}\left(\frac{\mathrm{d}p}{\mathrm{d}x} - \rho_\mathrm{l} g\right)\left(\frac{1}{2}y^2 - \delta y\right) + \frac{\tau_\mathrm{i}}{\mu_\mathrm{l}}y. \tag{4}$$

The $\tau_\mathrm{i}$ is the interfacial shear stress and is dependent upon the local mixture velocity, or

$$\tau_\mathrm{i} = \frac{1}{2}\rho u^2 f. \tag{5}$$

The condensing liquid flow rate would be obtained as

$$M = \rho_\mathrm{l}\pi d \int_0^\delta u_\mathrm{l}\,\mathrm{d}y. \tag{6}$$

To determine the heat transfer parameters $\delta$ and $t_\mathrm{i}$ must be known. These two parameters were connected with heat and mass transfer at the interface. At the interface the water vapor is at saturation, so the temperature $t_\mathrm{i}$ can be decided by the concentration of the water vapor $c_\mathrm{i}$. The determination of interfacial temperature $t_\mathrm{i}$ relies on the diffusion flow of non-condensable gas towards the interface in the mixture bulk and also the mass transfer coefficient along the interface should be known.

### 3.2. Mass transfer

The mass diffusion equation without source reads

$$\rho\left(\frac{\partial c}{\partial \tau} + u\frac{\partial c}{\partial x} + v\frac{\partial c}{\partial y}\right) = \rho D\left(\frac{\partial^2 c}{\partial x^2} + \frac{\partial^2 c}{\partial y^2}\right). \tag{7}$$

The transport process is restricted to a steady situation, so the time term is identically zero. The variation of $c$ is small enough for the terms $\partial/\partial x$ to be neglected as well. As a result, Eq. (7) reduces to

$$\rho v\frac{\partial c}{\partial y} = \rho D\frac{\partial^2 c}{\partial y^2}. \tag{8}$$

Considering the mass diffusion in $y$ direction, Fick's equation for water vapor can be written as

$$\rho_\mathrm{v}(v_\mathrm{v} - v) = -\rho D\frac{\mathrm{d}c}{\mathrm{d}y}. \tag{9}$$

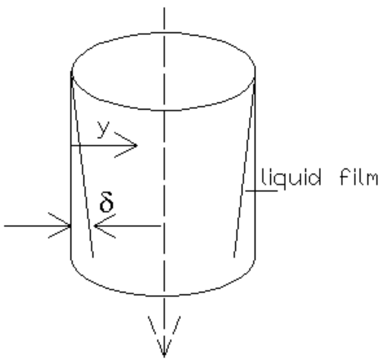

Fig. 3. Filmwise condensation in vertical tube.

When vapor fraction in the wet flue gas is defined as

$$c = \frac{\rho_v}{\rho} = \frac{\rho - \rho_n}{\rho}. \tag{10}$$

The mixture flux is

$$\rho v = \rho_v v_v + \rho_n v_n. \tag{11}$$

So the velocity is caused by condensation, $v$, is

$$v = -\frac{D}{1-c}\frac{\mathrm{d}c}{\mathrm{d}y}. \tag{12}$$

The distribution of vapor fraction near the interface can be derived combining Eqs. (9) and (12) with the suitable boundary condition [11].The mass transfer (condensing rate) at the interface can be derived as

$$m = -\rho v_i = \frac{\rho D}{1-c}\frac{\mathrm{d}c}{\mathrm{d}y}\bigg|_{y=0}. \tag{13}$$

Comparing with mass diffusion without condensation, the induced velocity intensifies the mass transfer at the interface. As the earliest investigator, Ackermann [12] studied this phenomenon and introduced the following expressions for modifying interfacial mass transfer:

$$\Theta_m = \frac{-\phi_m}{e^{-\phi_m} - 1}, \tag{14}$$

$$\phi_m = \frac{m}{g_m}, \tag{15}$$

where the coefficient, $m$, can be obtained from Eq. (13).

Accounting for the effect of condensation mass transfer at the interface should be modified as

$$sh = \Theta_m sh_0, \tag{16}$$

where $sh_0$ is the Sherwood number for single-phase convective mass transfer.

So mass transfer will be expressed as:

$$m = \Theta_m g_m \frac{c_g - c_i}{1 - c_i}. \tag{17}$$

### 3.3. Heat transfer

Accordingly, with the consideration of vapor condensation occurring at the interface the associated convection heat transfer between gas bulk and condensing liquid would be expressed as:

$$Nu = \Theta_t Nu_0. \tag{18}$$

The correct factor was given as [11]

$$\Theta_t = \frac{-\phi_t}{e^{-\phi_t} - 1}. \tag{19}$$

The thermal dimensionless mass flux in Eq. (19) is

$$\phi_t = \frac{m c_{p,v}}{h_{g0}} \tag{20}$$

So the convective heat transfer coefficient is

$$h_g = \Theta_t h_{g0}, \tag{21}$$

where $h_{g0}$ is the convective heat transfer coefficient for a single-phase gas.

Mizushina et al. [13] successfully employed the thermal correction factor in the research on heat transfer of pure vapor by accounting for condensation mass flux. However, nobody has extended this to the case of wet flue gas.

The associated condensation (or latent) heat transfer can be obtained from the account of mass transfer or condensed liquid determined by Eq. (17).

### 3.4. Analytical procedure

For the wet flue gas flowing in a vertical tube the mass reduction in the longitudinal direction can be neglected because of very low vapor fraction and condensing rate at the interface. But the condensation must be considered in the condensing liquid film.

Considering Eqs. (3) and (17),

$$\frac{\mathrm{d}M}{\mathrm{d}x} = \pi d \Theta_m g_m \frac{c_g - c_i}{1 - c_i}, \tag{22}$$

$$M(x + \mathrm{d}x) = M(x) + \pi d \Theta_m g_m \frac{c_g - c_i}{1 - c_i}\,\mathrm{d}x. \tag{23}$$

Since the change of bulk gas mass flux is negligible, the change of vapor fraction of the bulk flow can be simplified as

$$\frac{\mathrm{d}c_g}{\mathrm{d}x} = -\frac{4\Theta_m g_m}{d}\frac{c_g - c_i}{1 - c_i}. \tag{24}$$

As vapor fraction, mass transfer coefficient and Ackermann correct factor are dependent on $x$, Eq. (24) cannot be solved analytically. Another difficulty is that the vapor concentration near the interface is highly related with convective heat transfer. So the temperature profile must be calculated first.

From energy balance the variation of the bulk gas temperature $t_g$ along the flow direction is obtained as

$$\frac{\mathrm{d}t_g}{\mathrm{d}x}\rho u = -\frac{4}{d} h_g (t_g - t_i). \tag{25}$$

In Eq. (25) the contribution of vapor condensed was neglected. Similar to Eq. (23) the numerical method was used to obtain the temperature distribution along the $x$ direction.

In each step $\Delta x$ of the axial distance, there are two unknowns, liquid film thickness $\delta$ and interfacial temperature $t_i$. The iteration method was used to solve the heat and mass transfer equations. The interfacial temperature was assumed to be in a local saturated state and the vapor fraction at the interface can be obtained from the tabulated properties of the vapor corresponding to the local temperature $t_i$. So the convective heat and mass transfer can be calculated from Eqs. (17), (21) and (23). If Eq. (1) was not satisfied the new interfacial temperature $t_i$ was assumed until a suitable thermal balance was obtained. For the next step $\Delta x$, the new gas bulk temperature $t_g$ and vapor fraction $c_g$ were computed from Eqs. (24) and (23). So the heat mass transfer for next step $\Delta x$ would be started.

To explore the contribution of latent and sensible heat transfer at the interface, the energy transport equation at the interface was rewritten as

$$h_g^*(t_g - t_i) = h_g(t_g - t_i) + mL. \tag{26}$$

So the dimensionless expression was derived as:

$$Nu^* = Nu_g + Nu_c, \tag{27}$$

where

$$Nu^* = \frac{h_g^* d}{k_g}, \tag{28}$$

$$Nu_g = \frac{h_g d}{k_g}, \tag{29}$$

$$Nu_c = \frac{mLd}{k_g(t_g - t_i)}. \tag{30}$$

In Eq. (27) the first-term on the right-hand side represents the convective heat transfer between bulk gas mixture and liquid film surface and the second-term was the condensation at the interface. In the following section latent and sensible heat transfer for wet flue gas would be discussed.

## 4. Theoretical prediction

### 4.1. Property of wet flue gas

The condensation occurring in the wet flue gas is different compared with the condensation of vapor with small non-condensable gas. Fig. 4 shows the mass and volume fraction of water vapor in the wet flue gas exhausted from natural gas boiler at different excess air coefficients. The mass fraction of vapor in the vapor–gas mixture varies typically from 0.10 to 0.13.

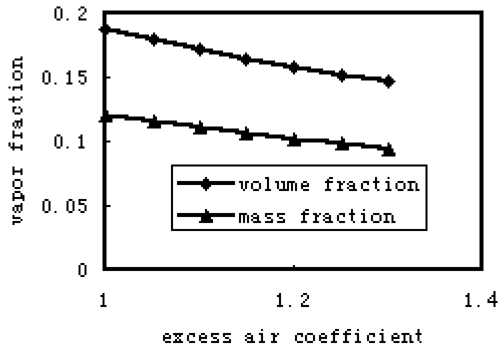


Fig. 4. The volume and mass fraction of vapor.

In the present situation the dew-point temperature of the wet flue gas (for convenience, wet flue gas was treated as vapor–air mixture) is lower than 60°C, referring to Fig. 5. When the wall temperature is higher than the gas dew-point, condensation will not occur, and the single-phase convective heat transfer governs the transport process. If wall temperature is lower than the corresponding dew-point temperature of the wet flue gas, condensation occurs and condensing rate is dependent on the wall temperature. Fig. 6 illustrates the maximum possible condensing rate of the water vapor in the flue gas (for cases of thermal equilibrium).

Accounting for the effect of temperature difference between gas bulk flow and the liquid film surface, the actual condensing rate would be largely lower than the value in Fig. 6. Apparently, the reduction of mass flux caused by condensation along the axial direction is negligible, referring to Eq. (24). Reasonably, the effects of velocity caused by condensation would become less important as the condensing rate decreases.

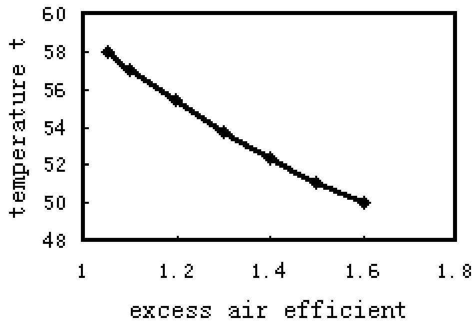


Fig. 5. The dew-point temperature of wet flue gas.

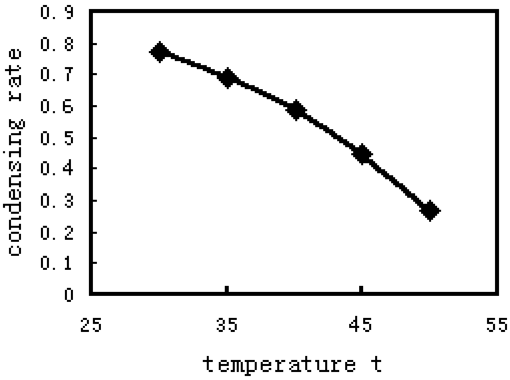


Fig. 6. The condensing rate at different wall temperatures.

### 4.2. The effect of condensation on convection heat transfer

Fig. 7 shows the correct factor values for convection heat transfer in the present research range. Very clearly, $\Theta_m \approx 1$, or the effects of the condensation at the interface can be neglected due to the wet flue gas in a vertical tube. So the sensible heat transfer can be predicted using single-phase correlation and not much error is caused.

For the wet flue gas flowing in a vertical tube the gas phase friction factor can be predicted as

$$f = 0.079Re^{-1/4}. \tag{31}$$

The gas convective heat transfer correlation in a vertical tube for the medium $Re$ number range was given by Von Volker Gnielinski [14]

$$Nu = 0.0214(Re^{0.8} - 100)Pr^{0.4}\left[1 + \left(\frac{d}{L}\right)^{2/3}\right]\left(\frac{T_f}{T_w}\right)^{0.45}. \tag{32}$$

Replacing the Nusselt number, $Nu$, in Eq. (32) with Sherwood number, $Sh$, and the Prandtl number $Pr$ by the Schmidt number, $Sc$, yields the following relation:

$$Sh = 0.0214(Re^{0.8} - 100)Pr^{0.4}Le^{0.4}\left[1 + \left(\frac{d}{L}\right)^{2/3}\right]\left(\frac{T_f}{T_w}\right)^{0.45}, \tag{33}$$

where the Lewis number, $Le = Sc/Pr$.

For vapor–gas mixture the Prandtl number is 0.7 and the Schmidt number is about 0.55, and the Lewis number, $Le$, is not equal to unity. As a result, $Sh$ will not exactly be equal to $Nu$. However, in the range of present tests, we have $Sh \approx Nu$ for $Re$ larger than 2000, Fig. 8 illustrates the $Nu$–$Re$ relation predicted by the model suggested.

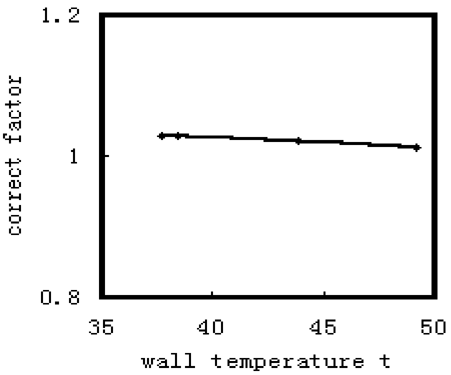


Fig. 7. The curve of the correct factor $\Theta_m$.

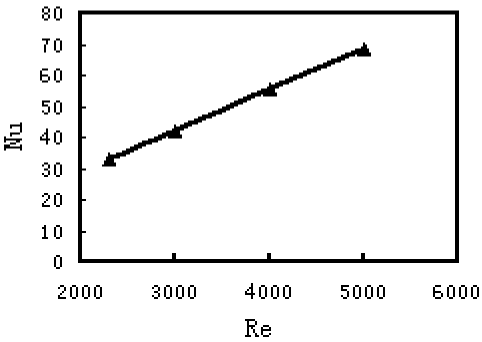


Fig. 8. $Nu$–$Re$ relation predicted by the developed model ($t_w = 37.79$°C).

### 4.3. Experimental results and discussion

A sequence of experiments was conducted to investigate the effect of condensation of vapor of the wet flue gas on the convection heat transfer in a vertical tube for designing the heat recovery device for the wet flue gas exhausted from a natural gas boiler. The experimental range was as same as that of practical industrial operation.

To minimize pressure drop in the gas passage and to be suitable for the actual condition, the Reynolds number, $u_g d/v_g$, of wet flue gas was in the transition region ($Re = 2300$–$5000$). Though some turbulent data for vapor–gas mixture were reported by some authors, most of the experimental results were in the range between laminar flow and turbulent flow while vapor condensation was lacking, especially for the low vapor fraction. The single-phase convection heat transfer cannot be neglected and basically dominates the transport process due to low vapor fraction in the wet flue gas. The role of condensation must be considered very carefully.

Fig. 9 illustrates the convective heat transfer results of wet flue gas without phase change and dry air convection in a tube. The heat transfer of wet flue gas is obviously higher than that of dry air. For wet flue gas in the experiments without condensation the correlation of convection heat transfer could be expressed in the following form as:

$$Nu = 0.0358(Re^{0.8} - 100)Pr^{0.4}\left[1 + \left(\frac{d}{L}\right)^{2/3}\right]\left(\frac{T_f}{T_w}\right)^{0.45}. \tag{34}$$

The curve for dry air was predicted by Eq. (32).

Although Eq. (32) includes the influence of $d/L$ and viscosity, they could be neglected in these situations because the corrector is approximated to unity. The coefficient of Eq. (34) is higher than that of the correlation (32). Two points should be noted. First, the wall

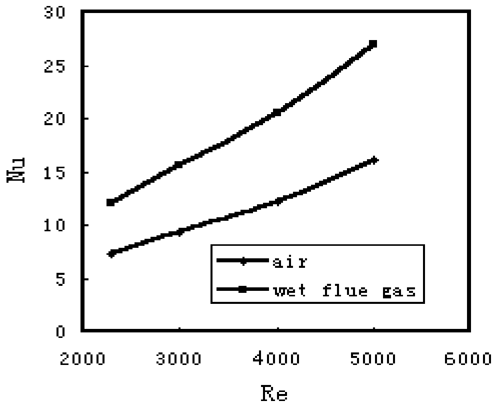


Fig. 9. The comparison of convection between air and flue gas.

temperature was used to replace interfacial temperature in Eq. (34). This would result in some decrease in *Nu*. Second, the actual dew-point temperature of wet flue gas is higher than that of the vapor–air mixture with same vapor concentration because of the existence of acid gas in the wet flue gas, and acid condensation or fog was formed. So the determination of the dew-point temperature of the wet flue gas should further be investigated in future.

Fig. 10 illustrates the *Nu–Re* relation at different wall temperature. Obviously, the wall temperature is an important parameter that alters the condensation contribution to the total heat transfer in a vertical tube. When wall temperature was high, bulk gas was in superheated situation. So the difference of vapor concentration between bulk and interface is less and mass transfer effect becomes less significance. Because of the lower mass fraction of vapor in the flue gas, the condensing film was much thin. The lower wall temperature produces lower vapor concentration at the interface of liquid film and induces larger concentration gradient in the wet flue gas when wall temperature was reduced. But the convection heat transfer between bulk flow and film or tube wall will still play an important role in the total heat transfer or dominate the transport process. Figs. 11 and 12 give the comparison of latent and sensible heat transfer component in the total heat transfer. These data were reduced by measuring the inlet and outlet temperatures of the bulk gas, bulk gas flow rate and condensing rate.

From Figs. 11 and 12, the latent contribution (condensation heat transfer) had about the same order as the sensible heat transfer (convection heat transfer) if the wall temperature was not very high. When the wall temperature was reduced, the latent heat transfer (condensation) reached a very great value and it could be over 80% (Fig. 12).

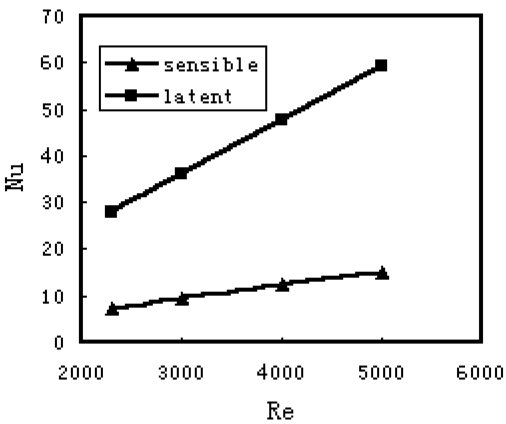


Fig. 11. The comparison between latent and sensible heat transfer ($t_w = 43.84°C$).

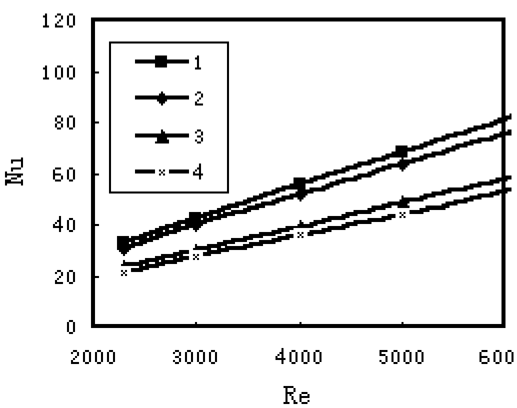


Fig. 10. The effect of wall temperature ((1) 37.74°C, (2) 38.4°C, (3) 43.84°C, (4) 49.17°C).

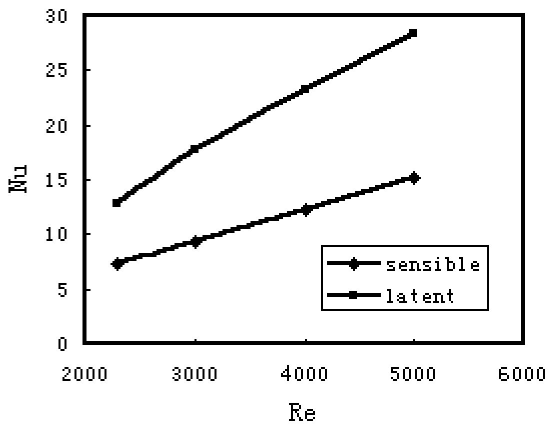


Fig. 12. The comparison between latent and sensible heat transfer ($t_w = 37.74°C$).

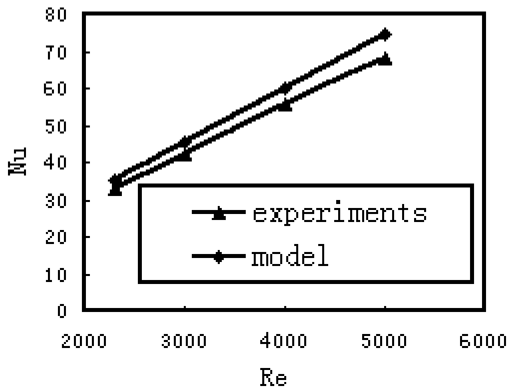


Fig. 13. The comparison between film model and experiments ($t_w = 37.79$°C).

The vapor fraction in the wet flue gas depends on the vapor partial pressure. When wet flue gas was in a superheated situation, the vapor concentration would not change when the bulk gas temperature increased or decreased. The mass transfer or condensation would depend on the vapor fraction at the interface for the given flowing condition. So the influence of superheated temperature difference on total heat transfer could be neglected, especially in the cases when the wall temperature was low (corresponding to great concentration gradient). The convection heat transfer would be increased because of the temperature difference between bulk flow and interface increasing. For the lower wall temperature the increment of the convection heat transfer can be neglected compared to the contribution of the condensation heat transfer.

Fig. 13 illustrates the predictions of the model developed by the present authors and experimental results clearly, are in quite good agreement.

In the vapor condensing process of wet flue gas, fog could be formed at some condition and has a strong influence on the condensation heat transfer [15]. Fog formation in the bulk or gas layer near to tube wall would alter the corresponding vapor concentration, and the drive force for mass transfer would be reduced. So condensation heat transfer is alleviated and convection heat transfer is increased because bulk temperature is increased. The effect of fog formation on total heat transfer would also depend on wall temperature.

## 5. Conclusion

1. The vapor condensation in the wet flue gas is an important factor and obviously enhanced convection heat transfer in a vertical tube.
2. The latent heat transfer (condensation heat transfer) contribution is dependent on the wall temperature and vapor concentration in the bulk flow. It could be as high as 80%. But the convection heat transfer cannot be neglected and plays a very important role, especially for cases of very low vapor fraction and high wall temperature (compared with the dew-point), in the wet flue gas.
3. The influence of excessive temperature difference on condensation heat transfer can be eliminated at lower wall temperature.
4. The fog formation may affect the total heat transfer, which depends on the wall temperature.


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